CHARACTERISTICS OF A HIGH-SPEED AXIAL BLOWER

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A THESIS

Submitted in partial fulfillment
of the requirements for the Degree
of Master of Science in Aeronautical Engineering

By

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Atlanta, Georgia
1940

Approved:

Acknowledgments

The author wishes to express his sincere appreciation to Professor Montgomery Knight who suggested the investigation, gave assistance in planning the attack on the problem, and made many helpful suggestions during its development; Professor A. M. Schwartz and Dr. R. H. Mills for their many helpful suggestions; the State Engineering Experiment Station without whose financial assistance the investigation could not have been undertaken; and Mr. M. M. Morrow of the Ford Motor Company who was instrumental in furnishing us with valuable equipment from the Ford Company.

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CHARACTERISTICS OF A HIGH-SPEED AXIAL BLOWER

Introduction

This paper deals with the experimental determination of the characteristics of a high-speed axial blower.

Those who have conducted previous investigations in this field (Refs. 1,2,3) have recognized the possibility of raising the efficiency of the axial flow fan by increasing the speed. Industrial applications, however, have not justified the structural and mechanical difficulties as well as the high noise level encountered as a result of high-speed operation. Hence, no previous investigations have been conducted at tip speeds greater than about 18,000 feet per minute.

When it became necessary to develop an axial type blower to be used in the Georgia Tech helicopter (Ref.5), it was felt that the increase in efficiency and the decrease in weight would more than compensate for the disadvantage of the high speed. Tests were, therefore, conducted on an axial type blower of high solidity and operating at tip speeds ranging from 9,400 to 33,000 feet per minute. This practically doubled the range of speeds over which similar tests have been conducted.

The horsepower required and the horsepower delivered were measured and the efficiency calculated for various blade angles up to and slightly beyond the point at which the blades stalled.

Blower and Apparatus

Fig.1 is a schematic drawing of the blower and the duct system showing the points at which measurements were made to determine the power output of the blower. Figs.2 and 3 are views of the complete test stand.

Power was supplied by an 85 HP V-8 engine mounted in bearings concentric with the crankshaft. This permitted measurement of the torque forces on a 50 pound platform scale graduated in increments of 1/100 of a pound.

The engine was connected to the blower through a gear box having a step-up ratio of 3.54 to 1. This permitted operating the blower at speeds up to 14,000 R.P.M. without exceeding the rated engine R.P.M. The gear box consisted of an automobile differential in which the spider gears had been locked, thus eliminating the differential action. The shaft, normally used as the axle, was connected to the engine by means of a universal joint. The blower was connected to the pinion shaft through a flexible coupling.

The rotor was composed of a dural disk (Figs.4 and 5) fourteen inches in diameter with thirty-six blades equally spaced around the periphery. The disk was split along a plane normal to the axis of rotation and the blades were clamped in sockets between the two halves by means of filister head screws. Loosening these screws permitted adjustment of the blade angles. Fig.4 shows the component parts of the rotor and the method of attaching the blades to the disk. Fig.5 shows the assembled rotor.

The blades shown in Fig.4 were used only in the preliminary tests. They were machined from 17ST aluminum alloy bar stock. The section used was a modified R.A.F.6 two inches long having a constant chord to facilitate machining. Before any satisfactory results had been obtained, fatigue cracks appeared at the roots of the blades. Subsequent vibration tests conducted on these blades indicated that this was caused by impulses from struts in the airstream a short distance downstream from the blades. In the operating range of speeds these impulses occurred at approximately the natural frequency of the blades.

Rather than redesign the whole unit it was decided to try to find a more suitable material from which to make the blades. The ideal material should be one having high internal damping, a relatively high allowable stress, low specific gravity, and be easy to fabricate. After considerable investigation a thermoplastic sold under the trade name of "Lucite" was tried. This material which had an allowable tensile strength of 8,000 to 10,000 pounds per square inch and a specific gravity of 1.8 seemed promising. It was easily molded under dry heat at a temperature of about 320°F. and 5,000 pounds per square inch pressure. As this method gave an easy means of making a large number of uniform blades, a more efficient tapered planform was used. The dimensions of the blade are given in Fig.6. Fig.7 shows two steps in the manufacture of the blade. The one at the right shows the blade before the flash was removed, the one on the left being the finished product. The white streak near the root of the finished blade is an imperfection due to molding. The transparency of this material rendered such flaws easily discernible.

The annular duct system used in the test is shown in Fig.1. The inner portion of the intake duct was a straight circular section having the same diameter as the rotor hub. The outer portion of the intake duct was bell-mouthed with a section of constant diameter starting one inch ahead of the blades. The blower rotor ran inside of a Micarta shroud ring, a maximum clearance of 0.006 inches being maintained between the ring and the blade tips.

The discharge duct consisted of two concentric circular sections, the outer portion having a constant diameter and the inner portion increasing in diameter from the blades to the annular discharge orifice. This orifice had an outside diameter of 18 inches (45.72 cm.) and an inside diameter of 16.25 inches (41.26 cm.) giving an annular discharge area of about 46.3 square inches (.034 sq.m.) compared with the area of about 100.5 square inches (.065 sq.m.) at the blades.

Radial clearance between the blades and the shroud ring was maintained by five streamlined struts. These connected the ring with the inner duct which in turn was centered on the blower shaft by means of a bearing. The struts were located with the leading edge 1-1/4 inches downstream from the center plane of the blade. A symmetric section was used having a chord of 1-1/2 inches and a maximum thickness of 1/2 of an inch. Fig.8 is a view looking into the intake duct with the blower and inner portion of the intake duct removed so that the streamline support struts and the annular discharge duct are shown.

A special pitot tube (Fig.9) was devised so that both circumferential and radial surveys of the discharge orifice could be made. Discussion of the calibration of this tube will be found in the Appendix.

A total head tube was placed at a point midway between two streamline struts and at approximately 1-1/2 inches downstream from the center line of the rotor. It was mounted in such a manner that surveys could be made along a radial line at this point.

Heads were measured in millimeters of water by means of well type manometers filled with distilled water.

Blower speeds were measured by a standard aircraft tachometer recalibrated for a six-to-one reduction ratio and checked by means of a strobotach.

Tests and Procedure

For these tests only the manometric efficiency (Ref.1) was considered. This efficiency is the ratio of the total energy of discharge from the blower rotor to the net input. All losses, therefore, were excluded except those in the intake duct and due to blade tip clearance. This necessitated determining the gear and windage losses of the disk.

No attempt was made to straighten out the rotational component of the velocity, since both the total head and the pitot tube were mounted so that they could be lined up with the resultant flow.

A series of tests were conducted holding the blade angles constant and varying the speed. These tests were made for blade angles from 15 degrees to 35 degrees at five degree intervals. In order to give an additional point near the stall an extra run was made at 32-1/2 degrees. For each blade angle the speed was varied from 2000 R.P.M. to 7000 R.P.M. in 1000 R.P.M. increments. This gave a corresponding range of tip velocities from 157 feet per second (47.9 m/sec) to 550 feet per second (143.8 m/sec).

For each run the following data were recorded:

Torque force.
Total head at discharge.
Static head at discharge.
Total head at the blade.
Revolutions per minute.
Barometric pressure.
Room temperature.

A series of circumferential surveys conducted at 15 degree intervals determined a discharge velocity coefficient to be applied to the mean velocity at the zero degree station. This coefficient was found to be independent of blade angle and speed.

The variation of the density of the manometer fluid was so small for the range of room temperature experienced during the tests that a single correction factor could be used.

The combined gear and windage losses were determined by running the blower rotor with plugs inserted in the blade sockets giving a smooth surface to the hub.

All data were reduced to standard conditions of temperature and pressure.

Reduction of Data

For each blade angle a preliminary survey was made across the jet along a radial line at the reference circumferential station. By means of the relation

$$V = \sqrt{\frac{2h_w}{\rho_a}} \tag{1}$$

the velocity at each station along the radial line was calculated.
Where:

V = velocity in meters per second,

$h_w = h_t - h_s$,

h_t = total head in mm. water,

h_s = static head in mm. water,

ρ_a = density of air = 1.25 Newton per cubic meter for standard air.

Plotting these values gave a velocity distribution curve (Fig.10) from which the mean velocity and the per cent of deviation of the mean velocity from the velocity at some point near the center of the jet could be found. Numerous tests showed that this value was independent of the blower speed as long as the blade angle was held constant. Therefore, it could be applied as a correction factor in determining the mean velocity at any speed for a given blade angle.

Applying all the correction factors the mean velocity of discharge now becomes

$$\bar{V} = C_1 \times C_2 \times C_3 \times C_4 \sqrt{\frac{2h_w}{\rho_a}} \qquad (2)$$

where:

$\bar{V}$ = mean velocity in meters/sec.,

C_1 = 1.09 V = correction factor for pitot tube,

C_2 = 0.99 V = correction factor for circumferential velocity variation with relation to zero degree,

C_3 = 0.9955 V = correction factor for variation of density of manometer fluid with room temperature, for room temperatures of 80 to 100 degrees F.,

C_4 = correction factor for deviation of velocity at a station along a radial line from the mean velocity along that line.

Since ρ_a for standard conditions is a constant, all these may be combined in the form

$$\bar{V} = K \sqrt{h_w} \quad .$$

The effective width of the jet was determined by the zero velocity points on the mean velocity curves. (Fig.10.) This width was divided equally on either side of the geometric midpoint of the jet and the effective jet area calculated. Then the quantity of air "Q" in meters per second was

$$Q = \bar{V}A \qquad (3)$$

A being the effective jet area in square meters.

The mean total head at the blades, h_t, was found in the same manner as the mean jet velocity. (Fig.16.)

Correcting the total head for temperature the net power output of the blower can now be calculated from the equation

$$HP_B = \frac{h_t' Q}{76.3} \tag{4}$$

where 76.3 is a constant in Kg m/second and h_t' is the mean total head in mm. of water corrected for the temperature.

The input horsepower to the blower is

$$HP_E = \frac{2\pi N \ell F'}{33000} \sqrt{\frac{\rho}{\rho_o}} \tag{5}$$

where

N = Engine speed in R.P.M.

$= \frac{N_B}{3.54}$

ℓ = Length of the torque arm

= 2.5 ft.

F' = Torque force, F, minus the gear and windage losses.

The ratio $\sqrt{\frac{\rho}{\rho_o}}$ is introduced to correct the engine horsepower to standard conditions.

Substituting the numerical values, Eq.(5) becomes

$$HP_E = 0.135 \times 10^{-3} N_B \; F' \sqrt{\frac{\rho}{\rho_o}} \; . \tag{6}$$

Values of HP_B and HP_E as computed from Eqs.(4) and (6) respectively were plotted on large scale log-log paper,(Figs.11 and 12). Since the resulting curves were straight lines, any discrepancies in the data or computations were easily discernible.

The manometric efficiency (Ref.1) was calculated from values of HP_E and HP_B taken from the curves for corresponding blade angles and speeds by the relation

$$\eta = \frac{HP_B}{HP_E} \quad . \tag{7}$$

The resulting values were plotted against R.P.M. for blade angles, α, of 15,20,25,30,32.5, and 35 degrees (Fig.13). Fig.14 shows curves of efficiency versus angle of attack plotted for speeds of 2000,3000,4000,5000, 6000,7000 R.P.M.

Values of the blade angle for maximum efficiency, $\alpha_{\eta\ max.}$, as indicated by the curves of η versus α (Fig.14) were plotted against R.P.M.

In order to have a basis of comparison with tests conducted previously by Marks and Weske (Ref.3) and Marks and Flint (Ref.4), the static head for various blade angles at 7000 R.P.M. was calculated.

Knowing the areas of the orifice and the blades and the velocity at the orifice, the velocity head at the blades, h_{w_1}, could be found by the relation

$$h_{w_1} = \frac{(V_2 \times \frac{A_2}{A_1})^2 \rho_0}{2} \tag{8}$$

where the subscript (1) denotes the conditions at the blades and the subscript (2) the conditions at the orifice.

Then:

$$h_{s_1} = h'_{t_1} - h_{w_1} \tag{9}$$

where h_{s_1} = Static head at the blades

and h'_{t_1} = Total head at the blades corrected for temperature.

Fig.17 presents these values graphically. Since the static head varies as the square of the tip speed, the values of the static head for maximum efficiency of each blower were calculated on the basis of a tip speed of 33,000 ft./min. corresponding to 7000 R.P.M.

Results of the present tests are presented in TABLES I to VI and Figs.10 to 16. TABLE VII compares the equivalent static heads at tip speeds of 33,000 ft./min. of the Georgia Tech blower with the blowers tested in Refs.2 and 3.

The distribution of points in Figs.11 and 12 indicates a maximum variation of the experimental data from a theoretical straight line of less than 1-1/2 per cent with most of the points deviating less than 1 per cent.

Discussion of Results

The curves of manometric efficiency versus R.P.M. (Fig.13) indicate that there is a definite increase in efficiency with an increase in speed. This effect is much more marked at the smaller blade angles; that is, the slope of the efficiency curve decreases as the blade angle increases. This is characteristic of curves for propellers operating at speeds below those for maximum efficiency.

Fig.13 shows that a maximum efficiency of 95.5 per cent was obtained at 7000 R.P.M. and blade angles of both 20° and 25°. This would seem to indicate that the optimum blade angle lies somewhere between these values.

This fact is substantiated by the curves of η versus α (Fig.14). These curves show a maximum efficiency of 97 per cent for a blade angle of 22-1/2 degrees and 7000 R.P.M. This efficiency was obtained at a greater static head for an equivalent tip speed than the static heads corresponding to the maximum efficiencies of the blowers tested in Refs. 2 and 3. The static heads of the present blower and those tested in Refs. 2 and 3 are compared in TABLE VII for equivalent tip speeds of 33,000 ft. per min.

The curve of the blade angle for maximum efficiency versus R.P.M. (Fig.15) shows that this angle tends to decrease slightly as the speed increases. The trend of this curve indicates that the angle may approach a constant value if the speed is carried high enough.

A further indication of conditions under which the blades were operating was obtained from the curves of the total head at the blade, h_t, measured at different radii. A comparison of these curves shows that stalling begins at a blade angle of about 25 degrees and at about 53 per cent of the span. As the blade angle is increased, the stall progresses farther in toward the hub to the 21 per cent point at 35 degrees. This is as it should be for an untwisted tapered blade.

The higher efficiency obtained in the present tests can be attributed to several factors which tend to reduce the drag of the section. Centrifugal force due to the high tip speeds, 33,000 ft.per min. maximum, probably tended to remove the boundary layer. Induced drag was reduced to a minimum by the small tip clearances maintained. Speeds at which these tests were run correspond to a range of Reynolds Numbers from 65,000 to 228,000. These values lie along the portion of the Reynolds Number curve over which the drag decreases very rapidly as the Reynolds Number increases.

All other variables being held constant, an increase in speed gives an increase in Reynolds Number and a corresponding increase in efficiency due to the decreased drag. This explains the fact that within the range of these tests, the efficiency curves (Fig.13) show a definite rise as the speed is increased. This indicates that it would be advantageous to go to higher Reynolds Numbers by increasing the speed still further. However, the fact that tip speeds must be kep below 800 to 900 ft.per sec. limits this increase, for in this region losses due to compressibility effect will more than offset the gain due to reduced drag. It has also been shown by previous tests (Refs.1,2, and 3) that for quantities of flow below those for maximum efficiency, the efficiency tends to decrease as the static pressure increases.

Conclusions

Within the limits of these tests the following conclusions can be arrived at:

1. The manometric efficiency increased with an increase in speed, a maximum value of 97 per cent being obtained at 7000 R.P.M. and a blade angle of 22-1/2 degrees.

2. Static heads obtained were proportionately higher when compared with results of tests by previous investigations.

3. These tests indicate that still higher efficiencies could be obtained at higher tip speeds.

4. This increase in efficiency as compared with earlier investigations is due in part to decreased drag as a result of operating at more *favorable Reynolds Numbers, to the probable removal of the boundary layer*

due to the high centrifugal force, and to the small blade tip clearances used.

5. For the condition of maximum efficiency the blade angle tends to decrease slightly as the tip speed is increased.

6. It should be noted that tests by previous investigators show that for quantities of flow below those for maximum efficiency, the efficiency tends to decrease as the static head increases. This will also be true of the present tests.

APPENDIX

APPENDIX

A. Variation of Torque Tare Force with Speed:

R.P.M.	f lbs.
2000	0.42
3000	0.51
4000	0.61
5000	0.72
6000	0.85
7000	1.01

B. Calibration of Special Pitot Tube:

The special pitot tube (Fig.9) was calibrated against a standard Prandtl tube in a 2 inch high-speed jet shown in Fig.19.

The tube head in the high-speed duct, h_t', was measured at the point A. Readings of h_t and h_s were taken with both a standard and the special pitot tube; h_t and h_s being the total head and the static head respectively. The subscript (1) was used to denote the values for the standard tube and the subscript (2) those for the special tube. The values of h_t and h_s were plotted against h_t'. Using the values taken from the curves the velocity heads h_w were determined where

$$h_{w_1} = h_{t_1} - h_{s_1} ,$$

$$h_{w_2} = h_{t_2} - h_{s_2} .$$

The correction factor in terms of the velocity head is

$$K_1 = \frac{h_{w_1}}{h_{w_2}} .$$

The following table gives these values:

h'_t	h_{w_1}	h_{w_2}	K_1
100	90	75	1.200
200	180	152	1.185
300	273	231	1.182
400	373	313	1.190
500	475	395	1.200
600	575	478	1.200

The mean value of 1.19 was used as the correction factor in all subsequent tests. Since this is in terms of the velocity head, the square root was used and applied directly to the velocity in the form

$$V_1 = 1.09\ V_2 .$$

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2. Marks, Lionel S. and Weske, John R.: "The Design and Performance of an Axial-Flow Fan." A.S.M.E. Transactions, Vol.56, No.11, November, 1934, pp.807-813.

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4. Weick, Fred E.: Aircraft Propeller Design. McGraw-Hill Book Company,Inc., 1930.

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TABLE I

$\alpha = 15^{\circ}$

R.P.M.	HP_B	HP_E	η Per cent
2000	0.243	0.363	66.8
3000	0.858	1.158	72.1
4000	2.170	2.705	75.2
5000	4.340	5.340	78.1
6000	7.570	9.11	81.0
7000	12.150	14.25	84.0

TABLE II

$\alpha = 20^{\circ}$

R.P.M.	HP_B	HP_E	η Per cent
2000	0.432	0.516	84.8
3000	1.499	1.699	87.6
4000	3.670	4.050	90.5
5000	7.190	7.840	92.1
6000	12.490	13.240	94.1
7000	19.850	20.88	95.5

TABLE III

$\alpha = 25^{\circ}$

R.P.M.	HP_B	HP_E	η Per cent
2000	0.674	0.740	88.0
3000	2.13	2.34	90.5
4000	5.06	5.50	92.5
5000	10.00	10.73	93.6
6000	17.30	18.30	95.3
7000	28.05	28.80	95.5

TABLE IV

$\alpha = 30^{\circ}$

R.P.M.	HP_B	HP_E	η Per cent
2000	0.625	0.805	79.0
3000	2.190	2.66	81.2
4000	5.310	6.28	83.1
5000	10.40	12.24	84.5
6000	17.72	20.68	85.8
7000	27.70	32.15	87.2

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TABLE V

$\alpha = 32.5^{\circ}$

R.P.M.	HP_B	HP_E	η Per cent
2000	0.632	0.807	77.0
3000	2.16	2.69	78.5
4000	5.21	6.49	80.6
5000	10.20	12.63	81.0
6000	17.72	21.70	81.5
7000	27.85	34.10	82.2

TABLE VI

$\alpha = 35^{\circ}$

R.P.M.	HP_B	HP_E	η Per cent
2000	0.624	0.833	72.5
3000	2.10	2.75	73.2
4000	5.21	6.78	74.5
5000	10.20	13.22	74.5
6000	17.00	22.40	75.5
7000	26.85	35.20	75.5

TABLE VII

Equivalent Static Heads of Three Axial Blowers[1]

	h_s in. H_2O
Georgia Tech Blower	20.4
Marks and Weske (Ref.3)	10.9
Marks and Flint (Ref.4)	16.8

[1] Tip speeds 33,000 ft.per min.

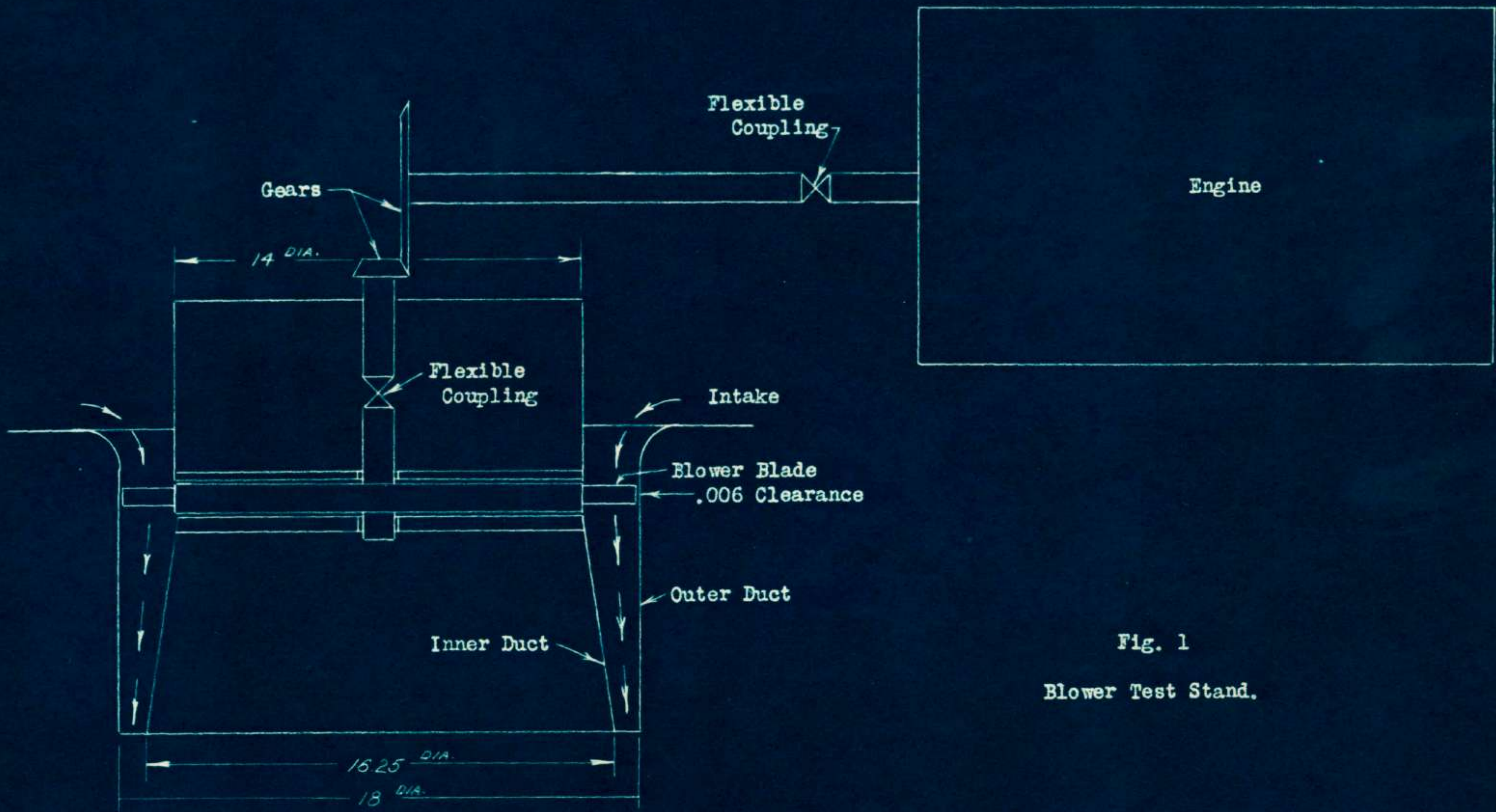


Fig. 1

Blower Test Stand.

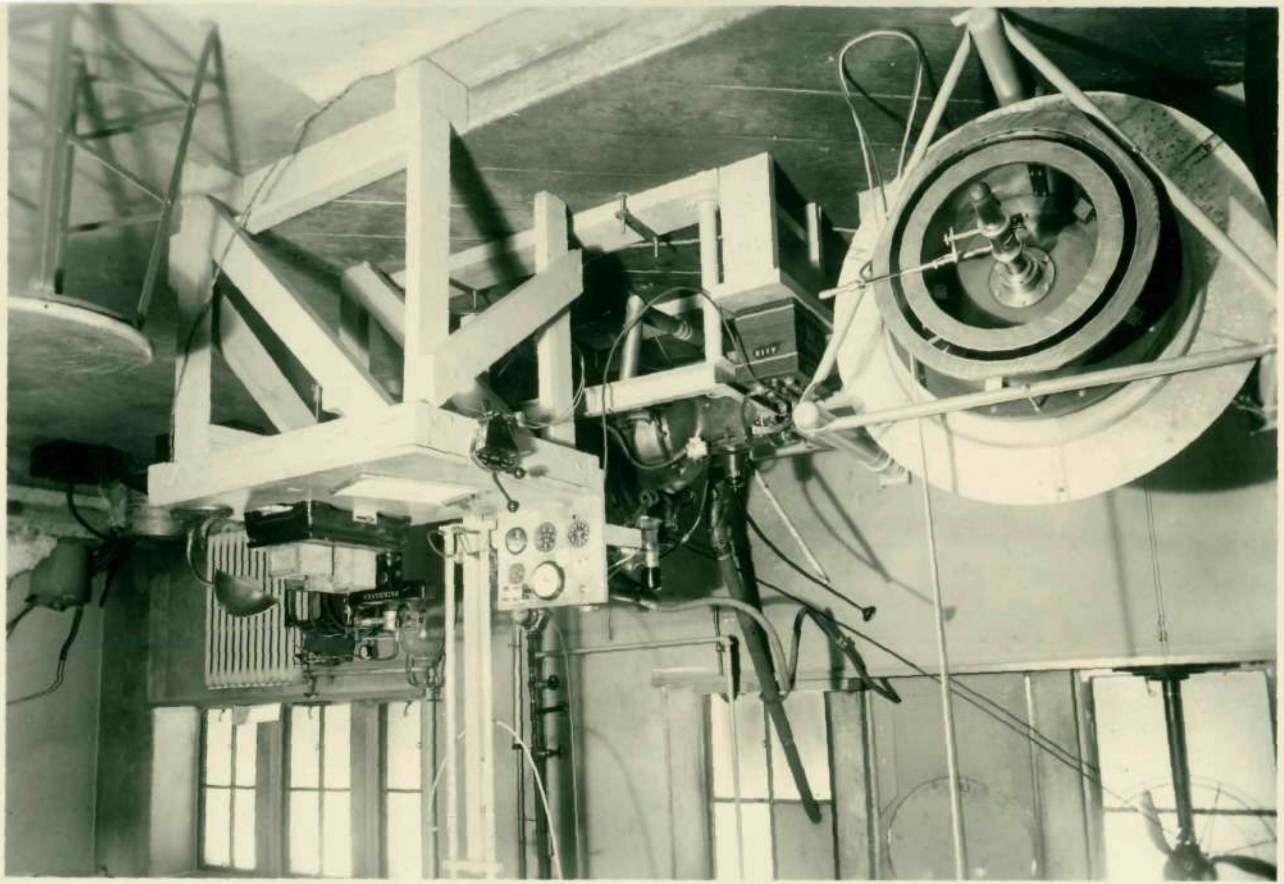


Fig. 2. General View of High-Speed Blower Laboratory Showing
Discharge Orifice and Control Table.

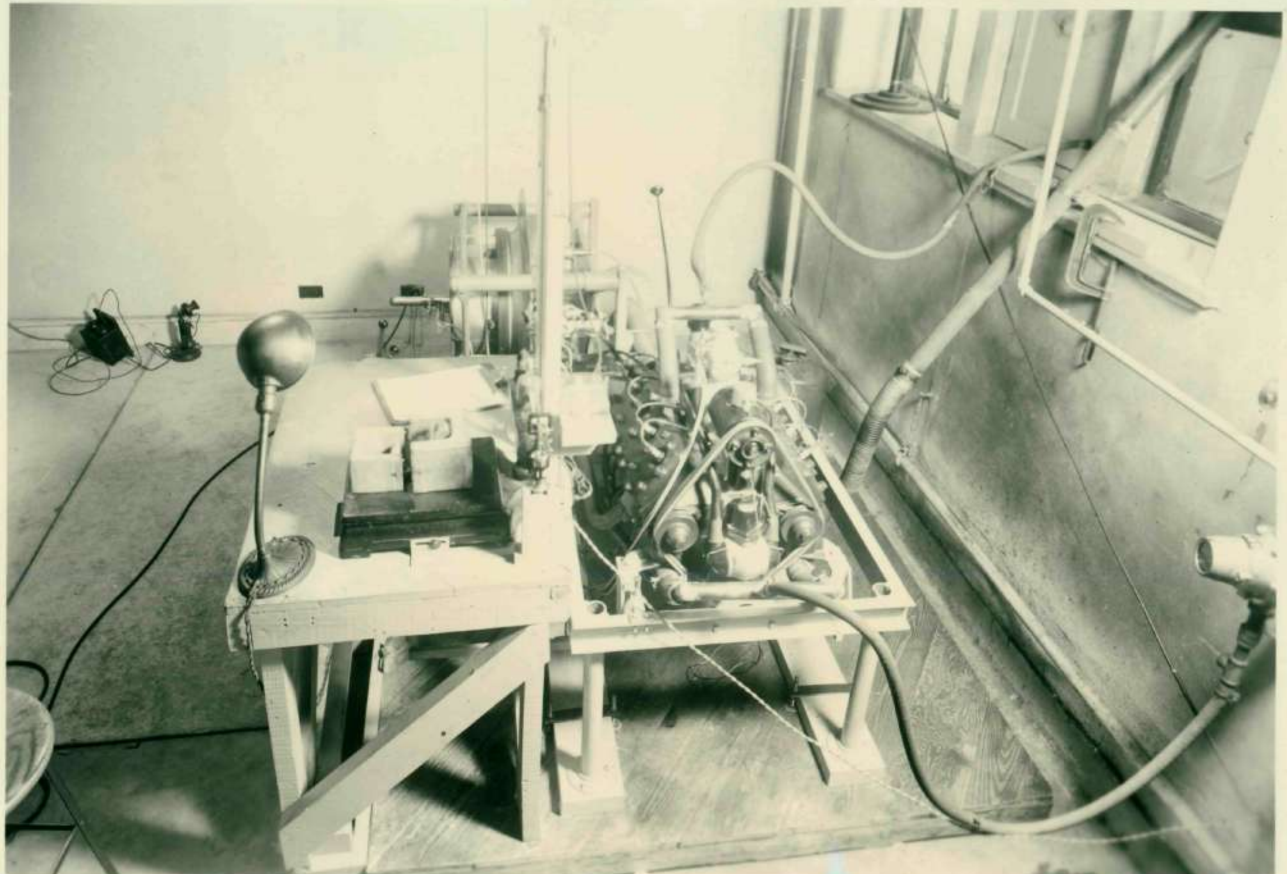


Fig. 3. General View of High-Speed Blower Laboratory
Showing Engine Installation.

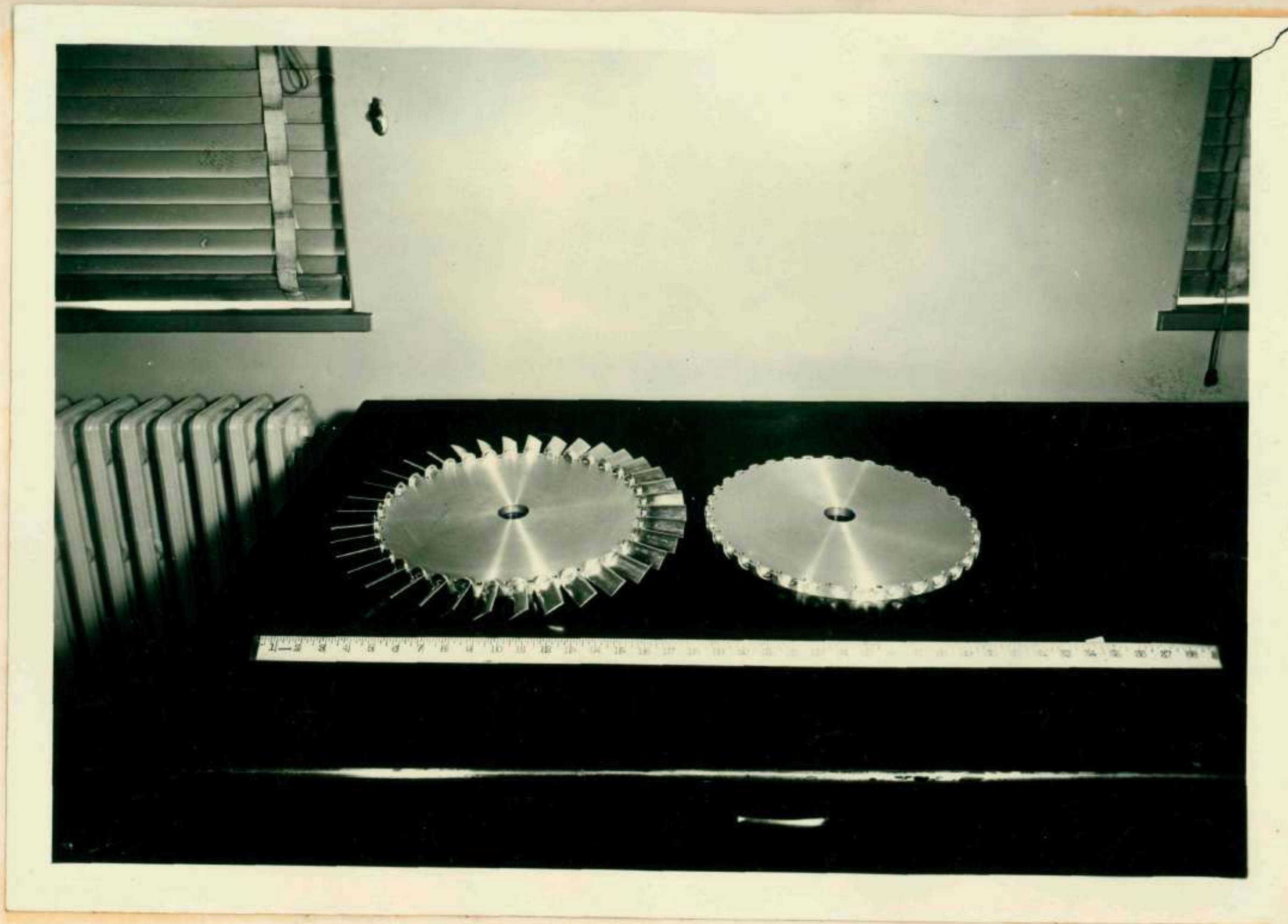


Fig.4. View Showing Component Parts of Blower.
(With Original Dural Blades)

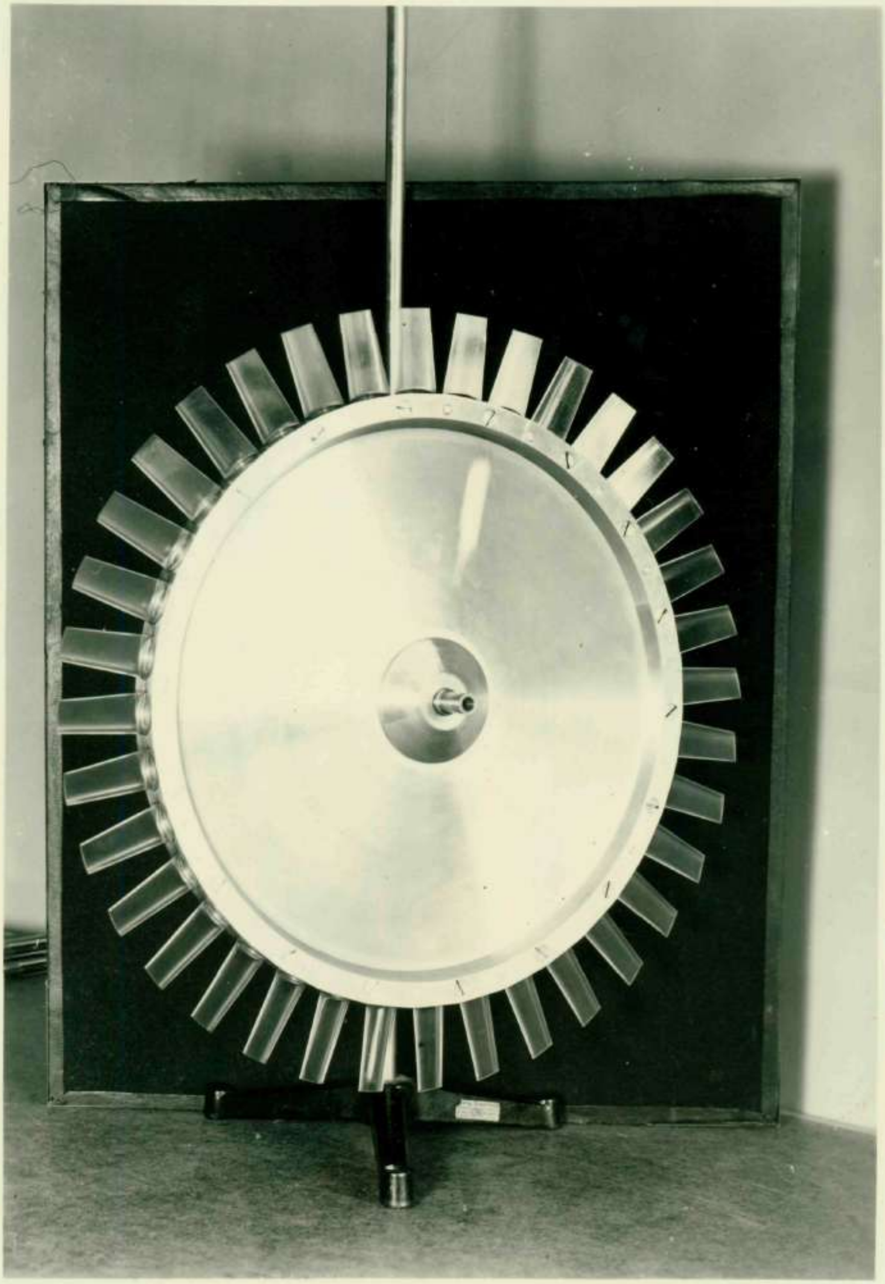
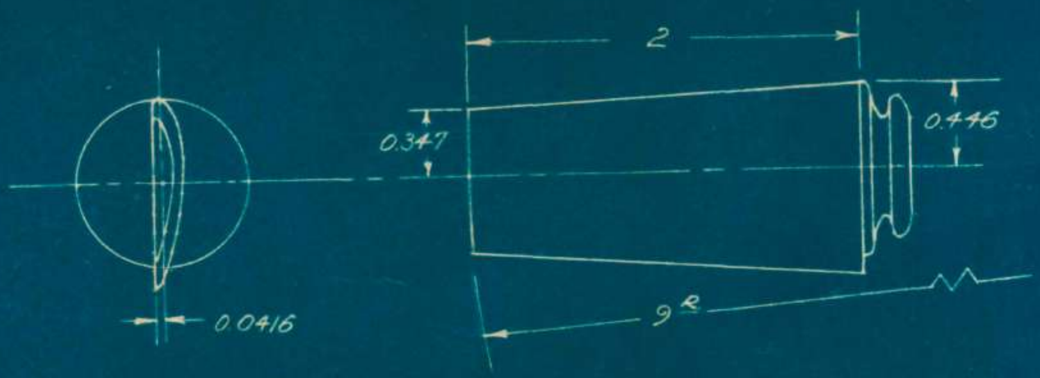


Fig. 5. High-Speed Axial Blower Rotor.
(With Final Lucite Blades)

Fig. 6. Principal Dimensions of Blades.



Blade Ordinates

Root

Distance from Leading Edge	Ordinate
L.E.Rad.	.015
.025	.062
.05	.089
.10	.119
.20	.143
.30	.150
.40	.148
.50	.143
.60	.131
.70	.111
.80	.084
.90	.053
T.E.Rad.	.012

Tip

Distance from Leading Edge	Ordinate
L.E.Rad.	.010
.019	.041
.037	.059
.075	.079
.150	.095
.225	.100
.300	.099
.375	.095
.450	.087
.525	.074
.600	.056
.675	.035
T.E.Rad.	.008

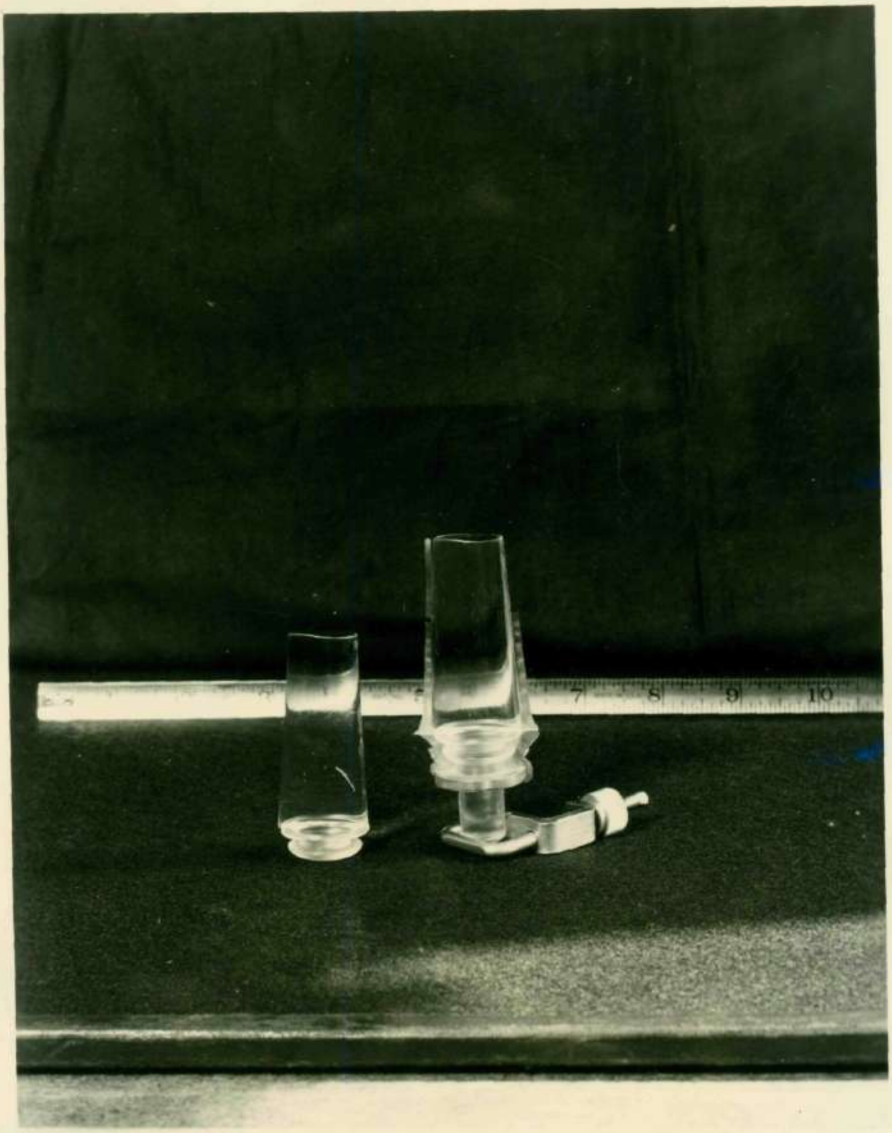


Fig. 7. Blade for High-Speed Axial Blower.

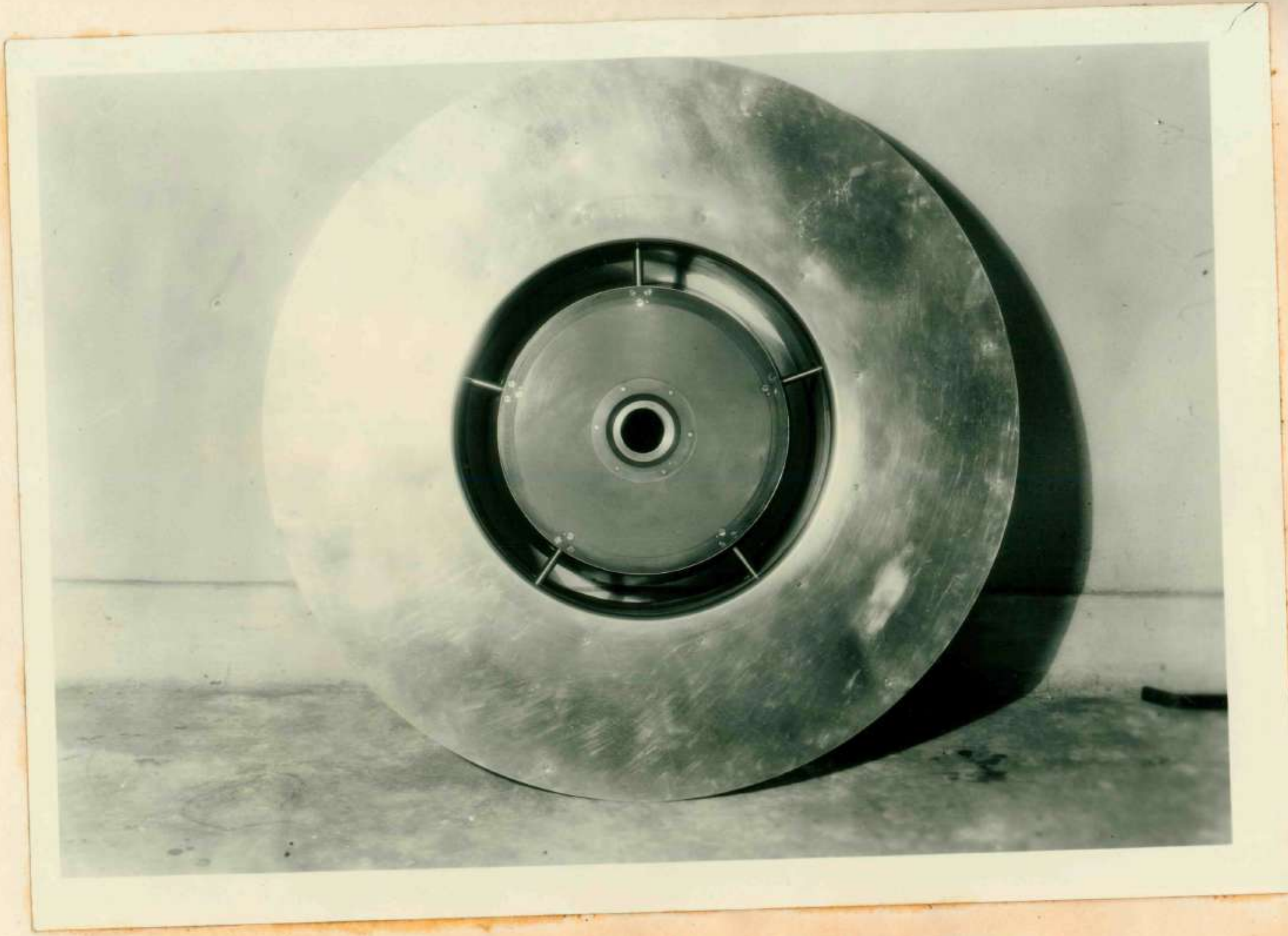


Fig. 8. Downstream View of Blower Duct with Blower and Inner Intake Duct Removed.

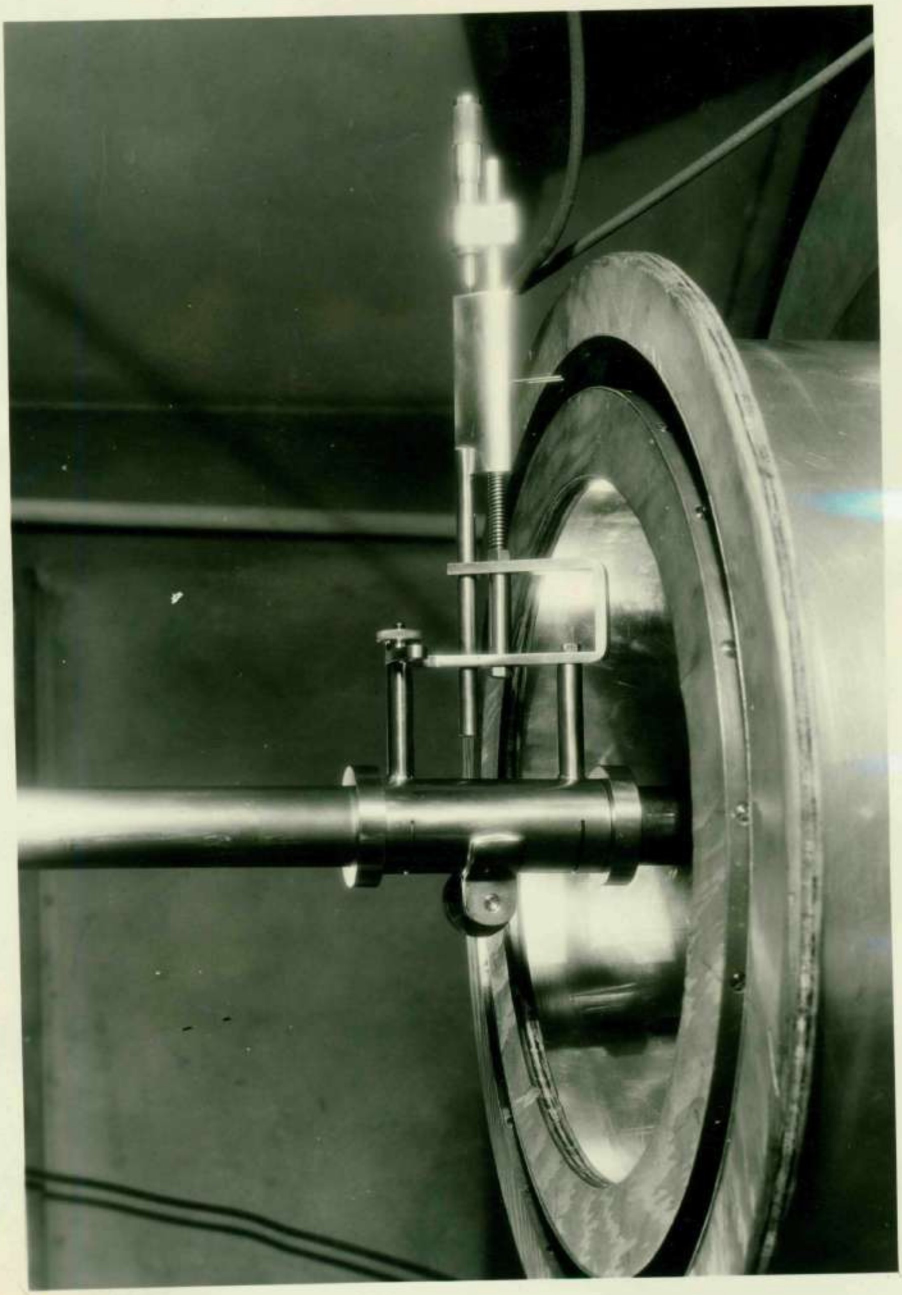


Fig. 9. View Showing Special Pitot Tube
and Method of Mounting.

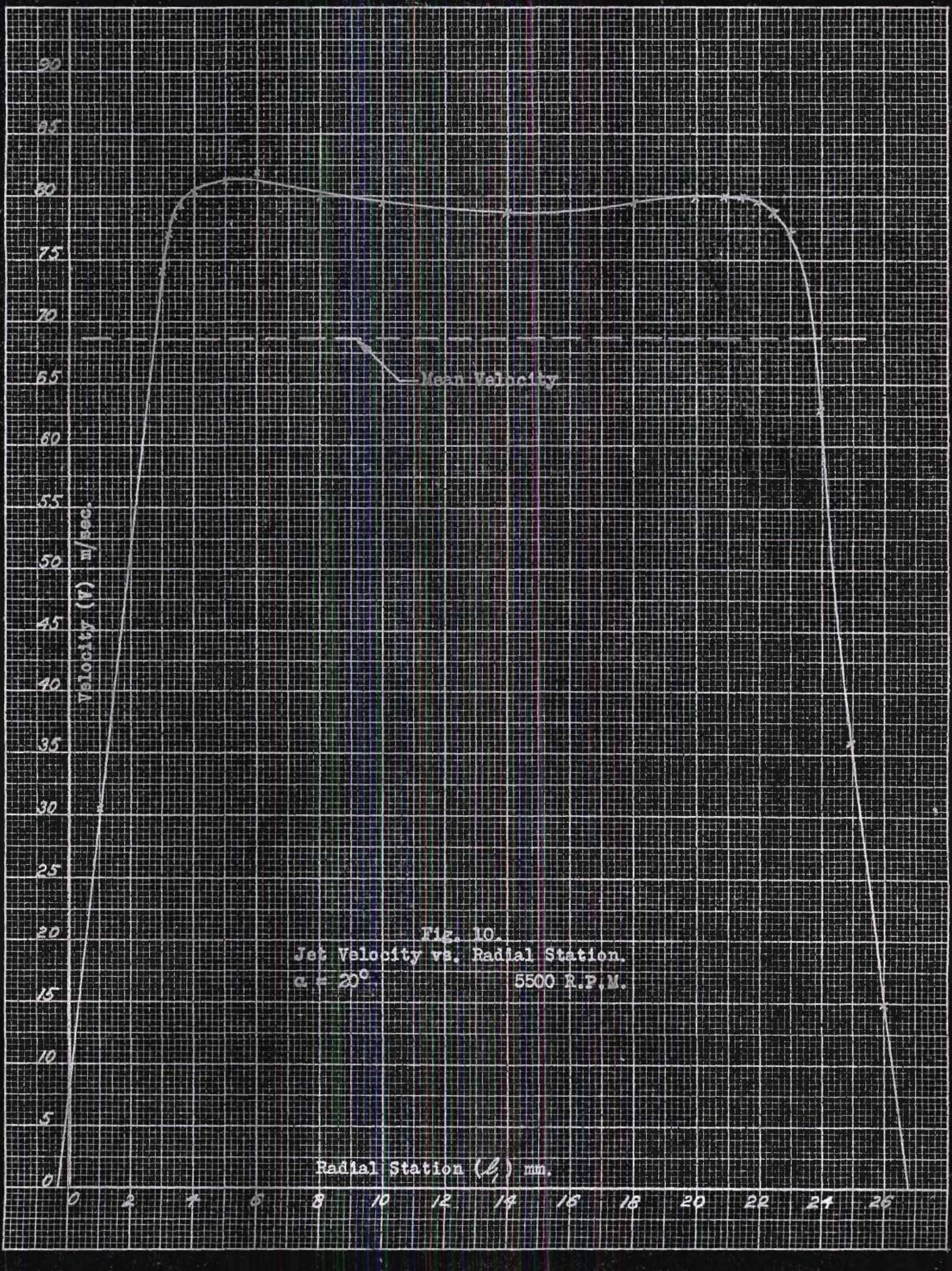
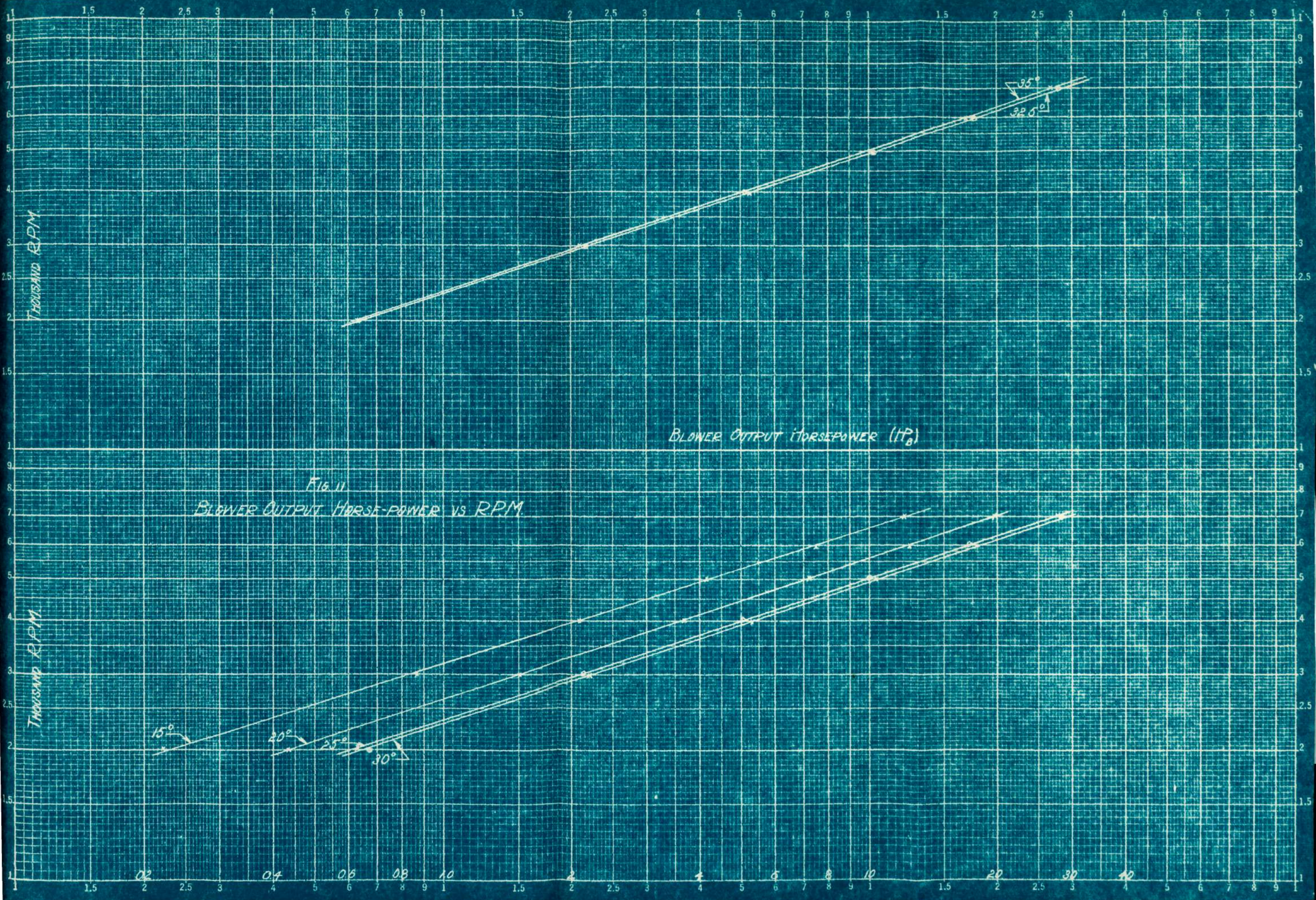


Fig. 10.
Jet Velocity vs. Radial Station.
$\alpha = 20°$. 5500 R.P.M.


Thousand R.P.M.
35°
32.5°
Blower Output Horsepower (HP_B)
Fig. 11
Blower Output Horse-power vs R.P.M.
Thousand R.P.M.
15°
20°
25°
30°
.02
.04
.06
.08
1.0
2
4
6
8
10
20
30
40

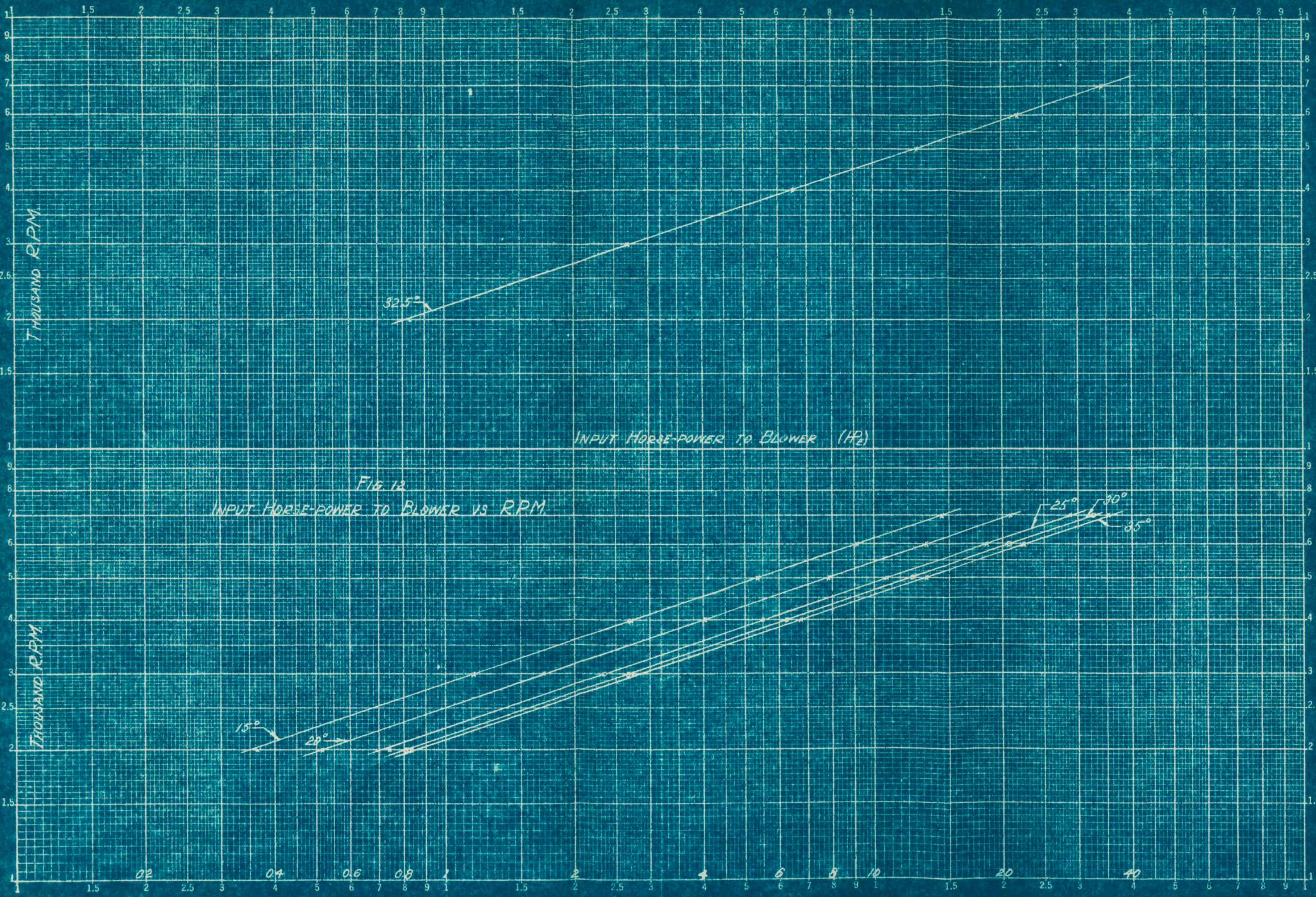


FIG. 12
INPUT HORSE-POWER TO BLOWER VS R.P.M.

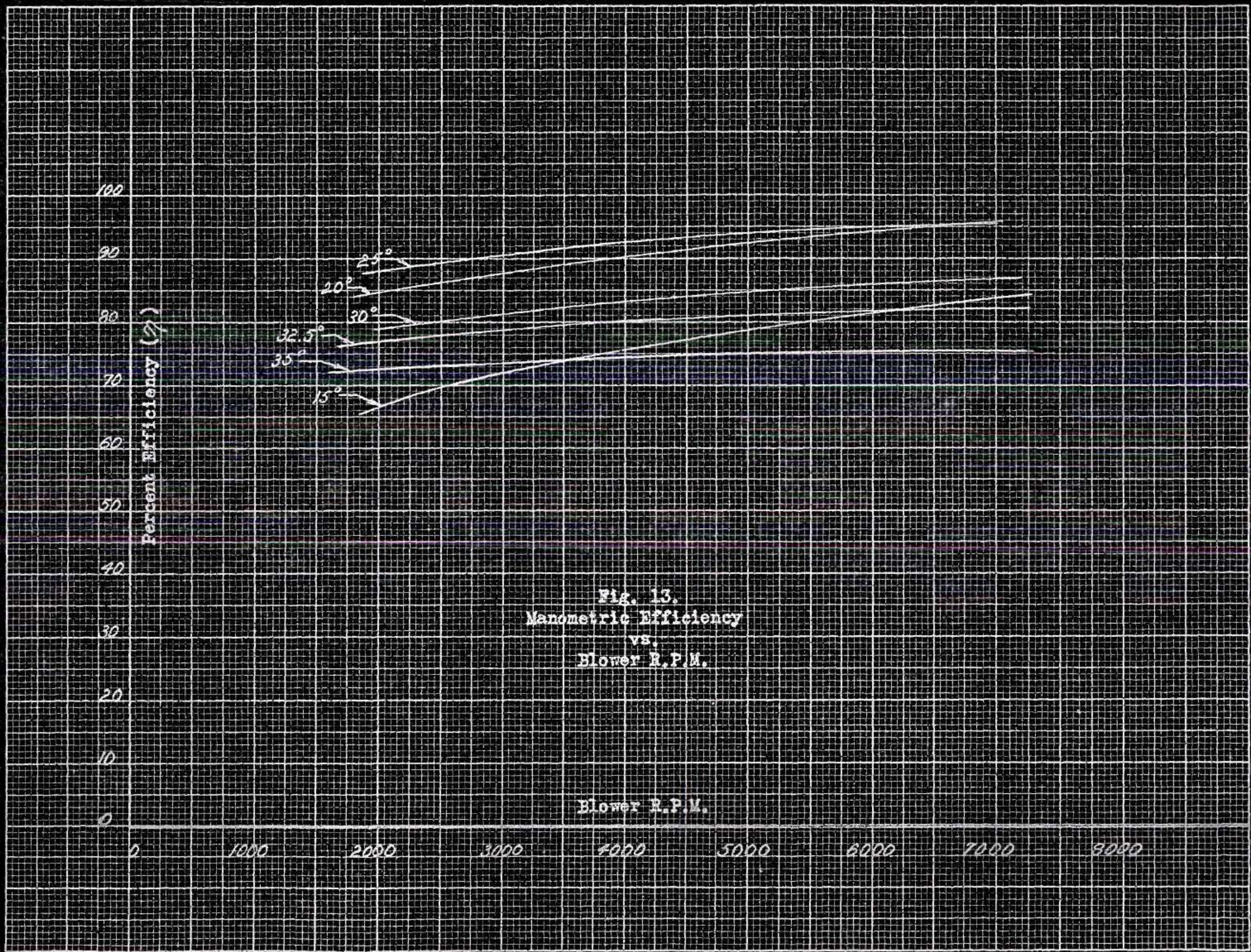


Fig. 13.
Manometric Efficiency
vs.
Blower R.P.M.
Percent Efficiency (η)
Blower R.P.M.
25°
20°
30°
32.5°
35°
15°
100
90
80
70
60
50
40
30
20
10
0
0
1000
2000
3000
4000
5000
6000
7000
8000

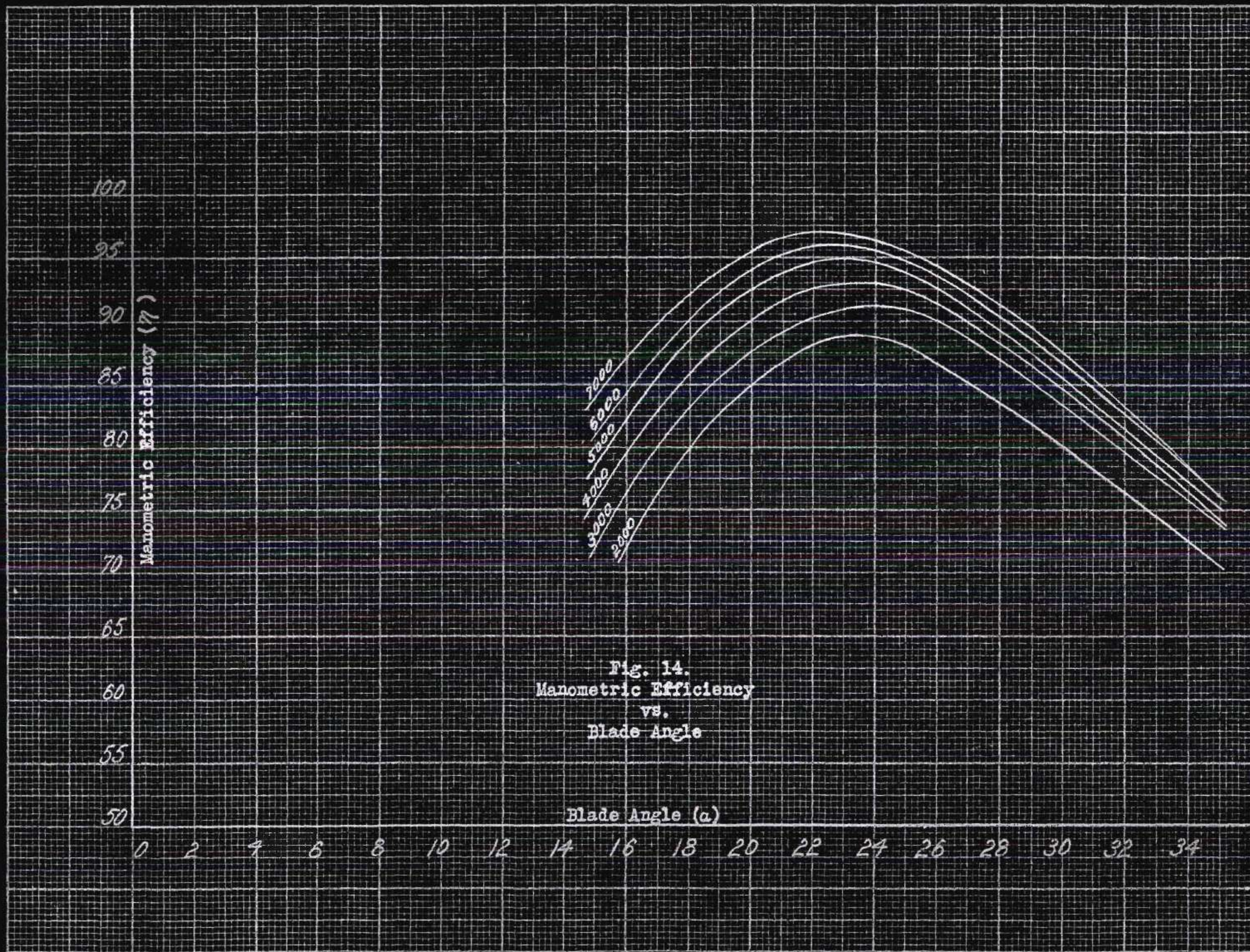


Fig. 14.
Manometric Efficiency
vs.
Blade Angle

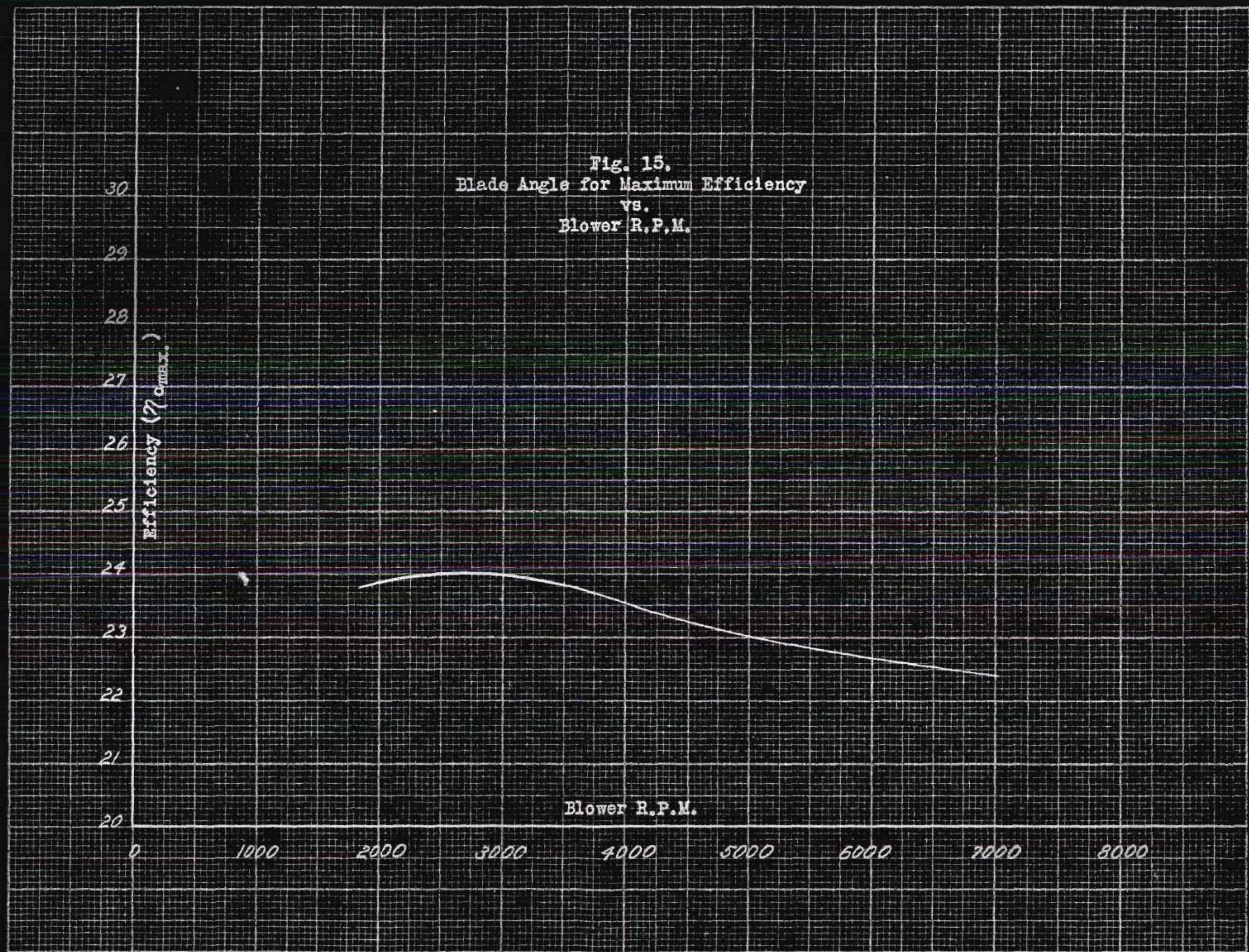


Fig. 15.
Blade Angle for Maximum Efficiency
vs.
Blower R.P.M.
Efficiency ($\eta_{o\,max.}$)
30
29
28
27
26
25
24
23
22
21
20
0
1000
2000
3000
4000
5000
6000
7000
8000
Blower R.P.M.

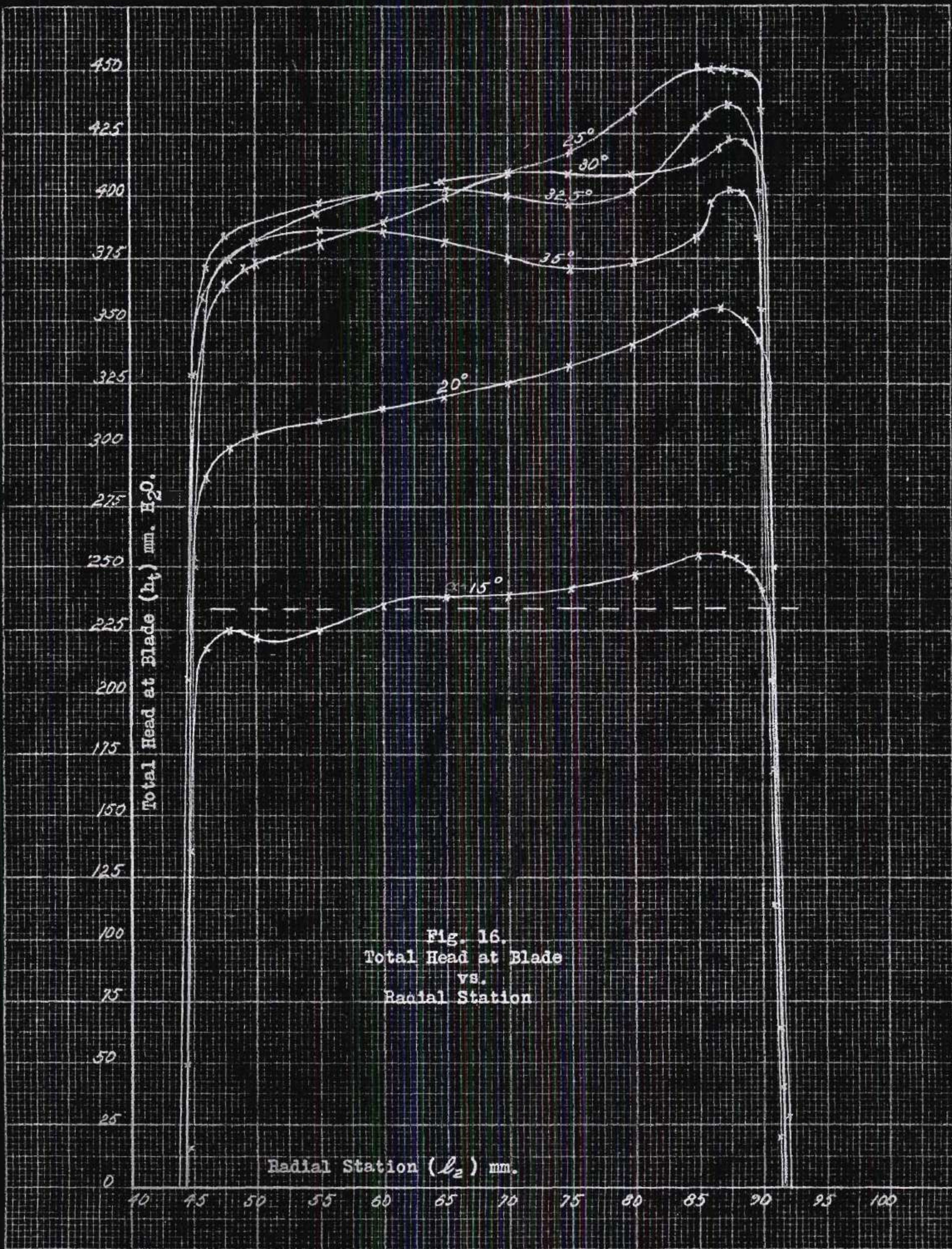


Fig. 16.
Total Head at Blade
vs.
Radial Station

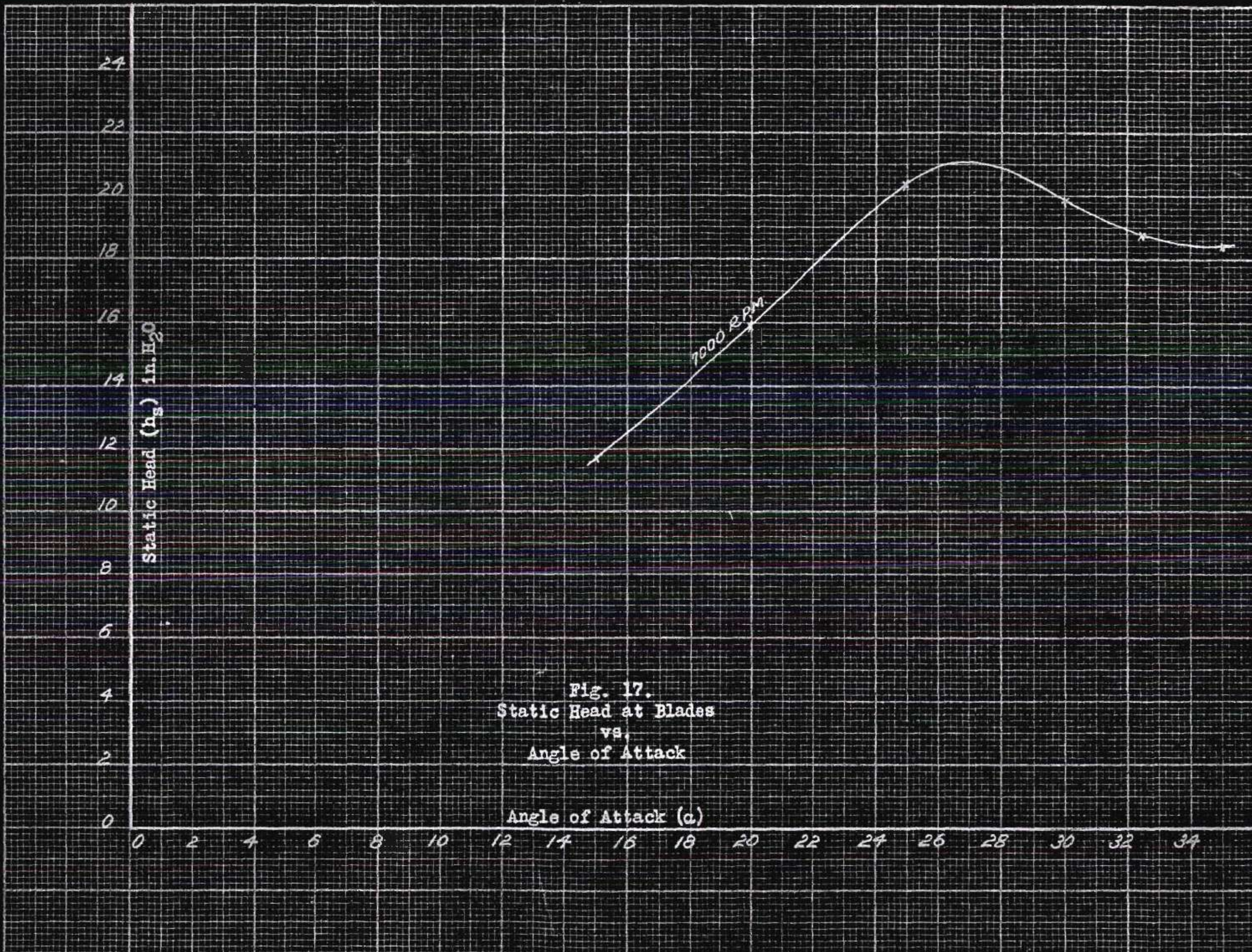


Fig. 17.
Static Head at Blades
vs.
Angle of Attack

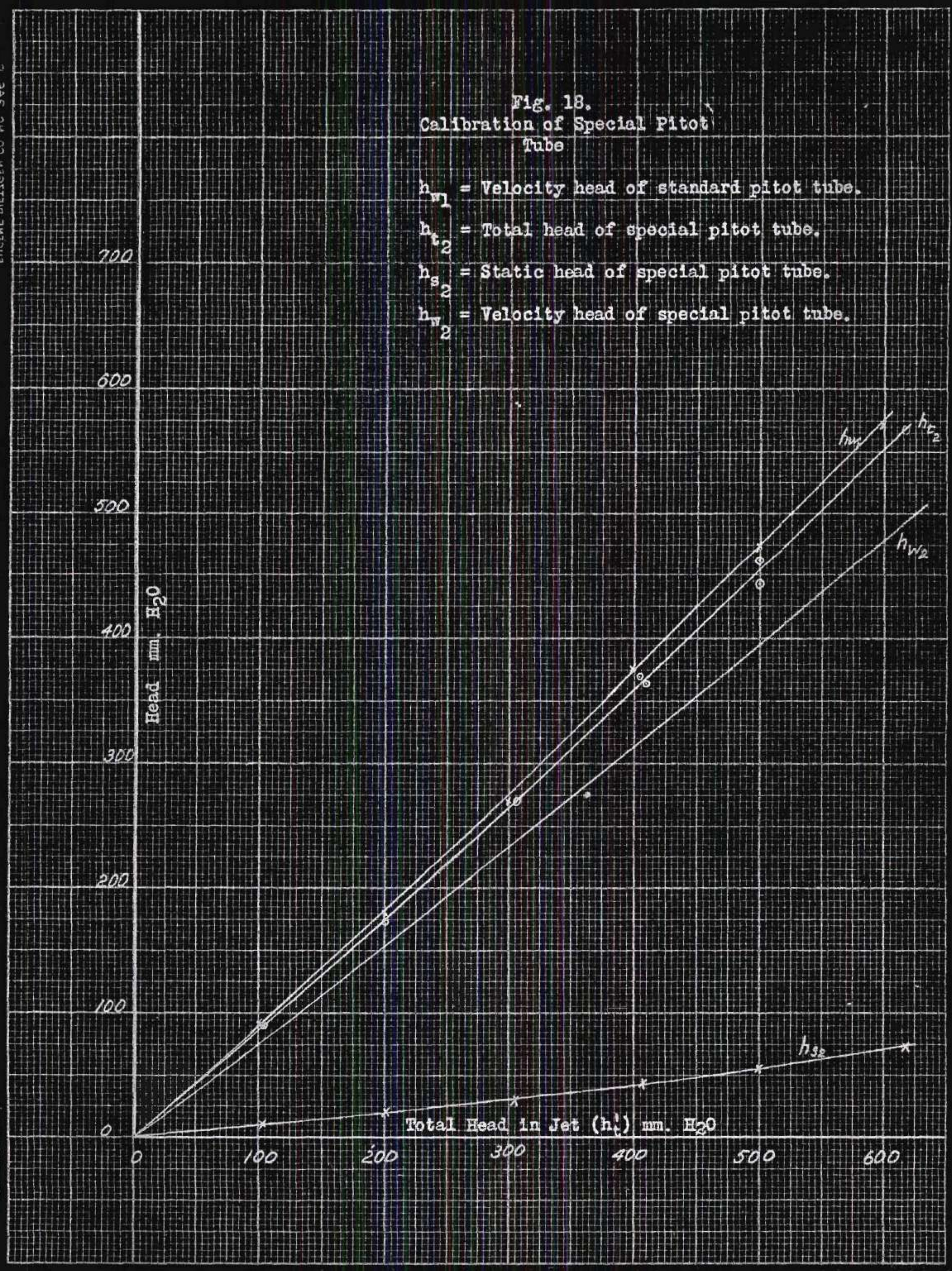


Fig. 18.
Calibration of Special Pitot Tube
h_{w_1} = Velocity head of standard pitot tube.
h_{t_2} = Total head of special pitot tube.
h_{s_2} = Static head of special pitot tube.
h_{w_2} = Velocity head of special pitot tube.
Head mm. H_2O
Total Head in Jet (h_t') mm. H_2O
0
100
200
300
400
500
600
700
h_{w_1}
h_{t_2}
h_{w_2}
h_{s_2}

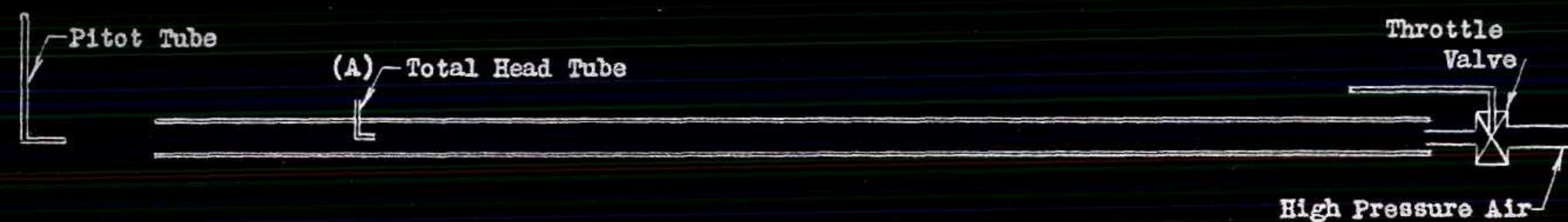


Fig. 19. High-Speed Jet for Pitot Tube Calibration.